

# **STRESS ANALYSIS OF THICK WALLED CYLINDER**

*A thesis*

*Submitted by*

**ROHIT KHETAN (109ME0377)**

*In partial fulfillment of the requirements  
For the award of the degree of*

**BACHELOR OF TECHNOLOGY  
in  
MECHANICAL ENGINEERING**

**Under the guidance of**

**Dr. H. ROY**



**Department of Mechanical Engineering  
National Institute of Technology Rourkela  
Odisha -769008, India**



### CERTIFICATE

This is to certify that this report entitled, “**Stress analysis of thick walled cylinder**” submitted by **Rohit Khetan (109ME0377)** in partial fulfillment of the requirement for the award of Bachelor of Technology Degree in Mechanical Engineering at National Institute of Technology, Rourkela is an authentic work carried out by them under my supervision.

To the best of my knowledge, the matter embodied in this report has not been submitted to any other university/institute for the award of any degree or diploma

Date:

Dr. H Roy  
Department of Mechanical Engineering  
(Research Guide)

## **ACKNOWLEDGEMENT**

I would like to give our deepest appreciation and gratitude to Prof. H Roy, for his invaluable guidance, constructive criticism and encouragement during the course of this project.

Grateful acknowledgement is made to all the staff and faculty members of Mechanical Engineering Department, National Institute of Technology, Rourkela for their encouragement. In spite of numerous citations above, the author accepts full responsibility for the content that follows.

Rohit Khetan

## **ABSTRACT**

It is proposed to conduct stress analysis of thick walled cylinder and composite tubes (Shrink fits) subjected to internal and external pressure. Many problems of practical importance are concerned with solids of revolution which are deformed symmetrically with respect to the axis of revolution. The examples of such solids are: circular cylinders subjected to uniform external and internal pressure. The stress analysis of thick walled cylinders with variable internal and external pressure is predicted from lame's formulae. Different case in lame's formula are thick walled cylinder having both (a) External and Internal pressure (b) Only Internal Pressure (c) Only External Pressure. In case of Composite tubes (Shrink Fit) the contact pressure between the two cylinders is determined then stress analysis is done by applying external and internal pressure in tube by lame's formulae. Theoretical formulae based results are obtained from MATLAB programs. The results are represented in form of graphs.

## **TABLE OF CONTENTS**

<b>TITLE</b>	<b>PAGE NO.</b>
CERTIFICATE	ii
ACKNOWLEDGEMENT	iii
ABSTRACT	iv
TABLE OF CONTENTS	v
LIST OF FIGURES	vii
NOTATIONS	viii

### **CHAPTER 1: INTRODUCTION**

1.1 Problem statement	1
1.2 Literature Review	2

### **CHAPTER 2: MATHEMATICAL MODELLING**

2.1 Lamé's Problem	3
2.1. a. Plane Stress	
2.1. a. i. Cylinder subjected to internal pressure only	6
2.1. a. ii. Cylinder subjected to external pressure only	6
2.2. b. Plane Strain	6

<b>TITLE</b>	<b>PAGE NO.</b>
<b>CHAPTER 3:RESULTS AND DISCUSSIONS</b>	
<b>3.1 Matlab Programmes</b>	
3.1.1 Thick Walled Cylinder	9
3.1.2 Shrink Fit	21
<b>CHAPTER 4: SUMMARY AND CONCLUSION</b>	
4.1 Summary	25
4.2 Future Scope of Work	25
REFERENCES	26

## LIST OF FIGURES

TITLE	PAGE NO.
Fig. 3.1: Graph between radial stress and radius for thick walled cylinder subjected to internal and external pressure	10
Fig. 3.2: Graph between hoop stress and radius for thick walled cylinder subjected to internal and external pressure	12
Fig 3.3:Graph between radial stress and radius for thick walled cylinder subjected to internal pressure only.	14
Fig. 3.4:Graph between hoop stress and radius for thick walled cylinder subjected to internal pressure only	16
Fig. 3.5: Graph between radial stress and radius for thick walled cylinder subjected to external pressure only	18
Fig. 3.6:Graph between hoop stress and radius for thick walled cylinder subjected to external pressure only	20
Fig. 3.7: Graph between radial stress and radius in case of shrink fit	22
Fig. 3.8:Graph between hoop stress and radius in case of shrink fit	24

## NOTATIONS

$\sigma_z$	Plane stress in z-axis
$\sigma_r$	Radial Stress
$\sigma_\theta$	Hoop Stress
$\tau_{rx}$	Shear Stress in rx-plane
$\tau_{ry}$	Shear Stress in ry-plane
$\tau_{rz}$	Shear Stress in rz-plane
$\varepsilon_z$	Strain in z-direction
$\varepsilon_\theta$	Circumferential strain
$\varepsilon_r$	Radial strain
E	Young's modulus
$\nu$	Poission ratio
$P_a$	Internal Pressure
$P_b$	External Pressure
$P_c$	Contact Pressure



# CHAPTER-1

## INTRODUCTION

### 1.1 Problem statement:

Thick walled cylinders are widely used in chemical, petroleum, military industries as well as in nuclear power plants. They are usually subjected to high pressure & temperatures which may be constant or cycling. Industrial problems often witness ductile fracture of materials due to some discontinuity in geometry or material characteristics. The conventional elastic analysis of thick walled cylinders to find radial & hoop stresses is applicable for the internal pressure up to yield strength of material.

General applications of Thick- Walled cylinders include, high pressure reactor vessels used in metallurgical operations, process plants, air compressor units, pneumatic reservoirs, hydraulic tanks, storage for gases like butane LPG etc.

In this Project we are going to analyze effect of internal and External Pressure on Thick walled cylinder, How radial stress & hoop Stress will vary with change of radius. Contact pressure in shrink Fit and its effect on hoop stress and radial stress is analysed.

## **LITERATURE REVIEW**

Xu & Yu [1] carried down shakedown analysis of internally pressurized thick walled cylinders, with material strength differences. Through elasto-plastic analysis, the solutions for loading stresses, residual stresses, elastic limit, plastic limit & shakedown limit of cylinder are derived.

Hojjati & Hossaini [2] studied the optimum autofriction pressure & optimum radius of the elastic plastic boundary of strain hardening cylinders in plane strain and plane stress conditions. They used both theoretical and Finite element modelling. Equivalent von-Mises stress is used as yield criterion.

M. Imanijed & G. Subhash [3] developed a generalized solution for small plastic deformation of thick-walled cylinders subjected to internal pressure and proportional loading.

Y.Z. Chen & X.Y. Lin [4] gave an alternative numerical solution of thick walled cylinder and spheres made of functionally graded materials.

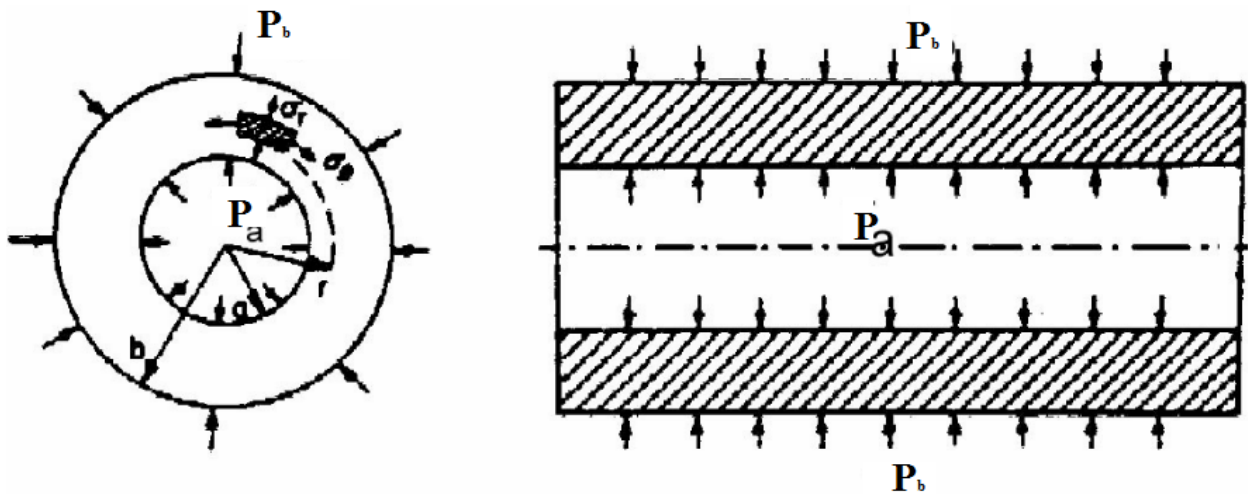
Li & Anbertin [5] presented analytical solution for evaluation of stresses around a cylinder excavation in an elastoplastic medium defined by closed yield surface.

## CHAPTER 2

### MATHEMATICAL MODELLING

#### 2.1 LAME'S PROBLEM-Thick walled cylinder subjected to internal and external pressure

Consider a cylinder of inner radius  $a$  and outer radius  $b$ . Let the cylinder to be subjected to internal pressure  $P_a$  and external pressure  $P_b$ . It will have two cases plane stress case ( $\sigma_z=0$ ) or as a plain strain case ( $\epsilon_z=0$ )



##### 2.1. a. Plane Stress

Let the ends of the cylinder be free to expand. We shall assume that  $\sigma_z=0$  our results just justify this assumption. Owing to uniform radial deformation  $\tau_{rz}=0$ , Neglecting body forces we can write

$$\frac{\partial \sigma_r}{\partial r} + \frac{\sigma_r - \sigma_\theta}{r} = 0$$

Since  $r$  is the only independent variable the above equation can be written as

$$\frac{d}{dr}(r\sigma_r) - \sigma_\theta = 0 \quad \text{-----eq(1)}$$

From Hooke's Law:

$$\varepsilon_r = \frac{1}{E}(\sigma_r - \nu \sigma_\theta)$$

$$\varepsilon_\theta = \frac{1}{E}(\sigma_\theta - \nu \sigma_r)$$

Stresses in terms of strain

$$\sigma_r = \frac{E}{1-\nu^2}(\varepsilon_r + \nu \varepsilon_\theta)$$

$$\sigma_\theta = \frac{E}{1-\nu^2}(\varepsilon_\theta + \nu \varepsilon_r)$$

After putting values  $\varepsilon_r$  and  $\varepsilon_\theta$

$$\sigma_r = \frac{E}{1-\nu^2} \left( \frac{du_r}{dr} + \nu \frac{u_r}{r} \right)$$

$$\sigma_\theta = \frac{E}{1-\nu^2} \left( \frac{u_r}{r} + \nu \frac{du_r}{dr} \right) \text{-----eq(2)}$$

Substituting above values in equation (1), we will get

$$\frac{d}{dr} \left( r \frac{du_r}{dr} + \nu u_r \right) - \left( \frac{u_r}{r} + \nu \frac{du_r}{dr} \right) = 0$$

$$\frac{du_r}{dr} + r \frac{d^2 u_r}{dr^2} + \nu \frac{du_r}{dr} - \frac{u_r}{r} - \nu \frac{du_r}{dr} = 0$$

$$\frac{d^2 u_r}{dr^2} + \frac{1}{r} \frac{du_r}{dr} - \frac{u_r}{r^2} = 0$$

$$\frac{d}{dr} \left[ \frac{1}{r} \frac{d}{dr} (u_r r) \right] = 0$$

$u_r$  can be found from this equation as

$$u_r = C_1 r + \frac{C_2}{r}$$

Substituting this values in Eq. (2)

$$\sigma_r = \frac{E}{1-\nu^2} \left[ C_1(1+\nu) - C_2(1-\nu) \frac{1}{r^2} \right]$$

$$\sigma_r = \frac{E}{1-\nu^2} \left[ C_1(1+\nu) + C_2(1-\nu) \frac{1}{r^2} \right]$$

$C_1$  and  $C_2$  are constants of integration and can be found out by applying boundary conditions.

When  $r=a$ ,

$$\sigma_r = -p_a$$

When  $r=b$ ,

$$\sigma_r = -p_b$$

so

$$\frac{E}{1-\nu^2} \left[ C_1(1+\nu) - C_2(1-\nu) \frac{1}{r^2} \right] = -p_a$$

$$\frac{E}{1-\nu^2} \left[ C_1(1+\nu) + C_2(1-\nu) \frac{1}{r^2} \right] = -p_b$$

On Solving,

$$C_1 = \frac{1-\nu}{E} \frac{p_a a^2 - p_b b^2}{b^2 - a^2}$$

$$C_2 = \frac{1+\nu}{E} \frac{a^2 - b^2}{b^2 - a^2} (p_a - p_b)$$

On substituting these values we get,

$$\sigma_r = \frac{p_a a^2 - p_b b^2}{b^2 - a^2} - \frac{a^2 b^2}{r^2} \frac{p_a - p_b}{b^2 - a^2}$$

$$\sigma_\theta = \frac{p_a a^2 - p_b b^2}{b^2 - a^2} + \frac{a^2 b^2}{r^2} \frac{p_a - p_b}{b^2 - a^2}$$

### 2.1. a. i. Cylinder Subjected to Internal Pressure only

In this case  $p_b=0$  and  $p_a=p$ .

Hence,

$$\sigma_r = \frac{p a^2}{b^2 - a^2} \left( 1 - \frac{b^2}{r^2} \right)$$

$$\sigma_\theta = \frac{p a^2}{b^2 - a^2} \left( 1 + \frac{b^2}{r^2} \right)$$

These equations show that  $\sigma_r$  is always a compressive stress and  $\sigma_\theta$  is a tensile stress.

### 2.1. a. ii. Cylinder subjected to external pressure only

In this case  $p_a=0$  and  $p_b=p$

Hence,

$$\sigma_r = -\frac{p b^2}{b^2 - a^2} \left( 1 - \frac{a^2}{r^2} \right)$$

$$\sigma_\theta = \frac{p b^2}{b^2 - a^2} \left( 1 + \frac{a^2}{r^2} \right)$$

### 2.1. b. Plain Strain

For long cylinder stresses are calculated as state of plane strain.

$\sigma_z$  presumed does not vary along the z axis.

$$\frac{d}{dr}(r\sigma_r) - \sigma_\theta = 0 \quad \text{-----eq(3)}$$

From Hooke's law

$$\varepsilon_r = \frac{1}{E} [\sigma_r - \nu(\sigma_\theta + \sigma_z)]$$

$$\varepsilon_\theta = \frac{1}{E} [\sigma_\theta - \nu(\sigma_r + \sigma_z)]$$

$$\varepsilon_z = \frac{1}{E} [\sigma_z - \nu(\sigma_r + \sigma_\theta)]$$

As  $\varepsilon_z=0$

$$\sigma_z = \nu(\sigma_r + \sigma_\theta)$$

$$\varepsilon_r = \frac{1+\nu}{E} [(1-\nu)\sigma_r - \nu\sigma_\theta]$$

$$\varepsilon_\theta = \frac{1+\nu}{E} [(1-\nu)\sigma_\theta - \nu\sigma_r]$$

While solving  $\sigma_r$  and  $\sigma_\theta$

$$\sigma_\theta = \frac{E}{(1-2\nu)(1+\nu)} [\nu\varepsilon_r + (1-\nu)\varepsilon_\theta]$$

$$\sigma_r = \frac{E}{(1-2\nu)(1+\nu)} [(1-\nu)\varepsilon_r + \nu\varepsilon_\theta]$$

Putting values of  $\varepsilon_r$  and  $\varepsilon_\theta$

$$\sigma_\theta = \frac{E}{(1-2\nu)(1+\nu)} \left[ \nu \frac{du_r}{dr} + (1-\nu) \frac{u_r}{r} \right]$$

$$\sigma_r = \frac{E}{(1-2\nu)(1+\nu)} \left[ (1-\nu) \frac{du_r}{dr} + \nu \frac{u_r}{r} \right] \text{-----eq(4)}$$

Substituting these in the equation of equilibrium

$$\frac{d}{dr} \left[ (1-\nu)r \frac{du_r}{dr} + \nu u_r \right] - \nu \frac{du_r}{dr} - (1-\nu) \frac{u_r}{r} = 0$$

$$\frac{du_r}{dr} + r \frac{d^2 u_r}{dr^2} - \frac{u_r}{r} = 0$$

$$\frac{d}{dr} \left( \frac{du}{dr} + \frac{u_r}{r} \right) = 0$$

We can write

$$u_r = C_1 r + \frac{C_2}{r}$$

Putting these values in equation(4)

$$\sigma_{\theta} = \frac{E}{(1-2\nu)(1+\nu)} \left[ C_1 + (1-2\nu) \frac{C_2}{r^2} \right]$$

$$\sigma_r = \frac{E}{(1-2\nu)(1+\nu)} \left[ C_1 - (1-2\nu) \frac{C_2}{r^2} \right]$$

Boundary conditions

When  $r=a$ ,

$$\sigma_r = -p_a$$

When  $r=b$ ,

$$\sigma_r = -p_b,$$

We can get

$$\frac{E}{(1-2\nu)(1+\nu)} \left[ C_1 - (1-2\nu) \frac{C_2}{a^2} \right] = -p_a$$

$$\frac{E}{(1-2\nu)(1+\nu)} \left[ C_1 + (1-2\nu) \frac{C_2}{b^2} \right] = -p_b$$

and solving we can find out

$$C_1 = \frac{(1-2\nu)(1+\nu)}{E} \frac{p_b b^2 - p_a a^2}{a^2 - b^2}$$

$$C_2 = \frac{1+\nu}{E} \frac{(p_b - p_a) a^2 b^2}{a^2 - b^2}$$

Substituting these values we can find out that

$$\sigma_r = \frac{p_a a^2 - p_b b^2}{b^2 - a^2} - \frac{a^2 b^2}{r^2} \frac{p_a - p_b}{b^2 - a^2}$$

$$\sigma_{\theta} = \frac{p_a a^2 - p_b b^2}{b^2 - a^2} + \frac{a^2 b^2}{r^2} \frac{p_a - p_b}{b^2 - a^2}$$

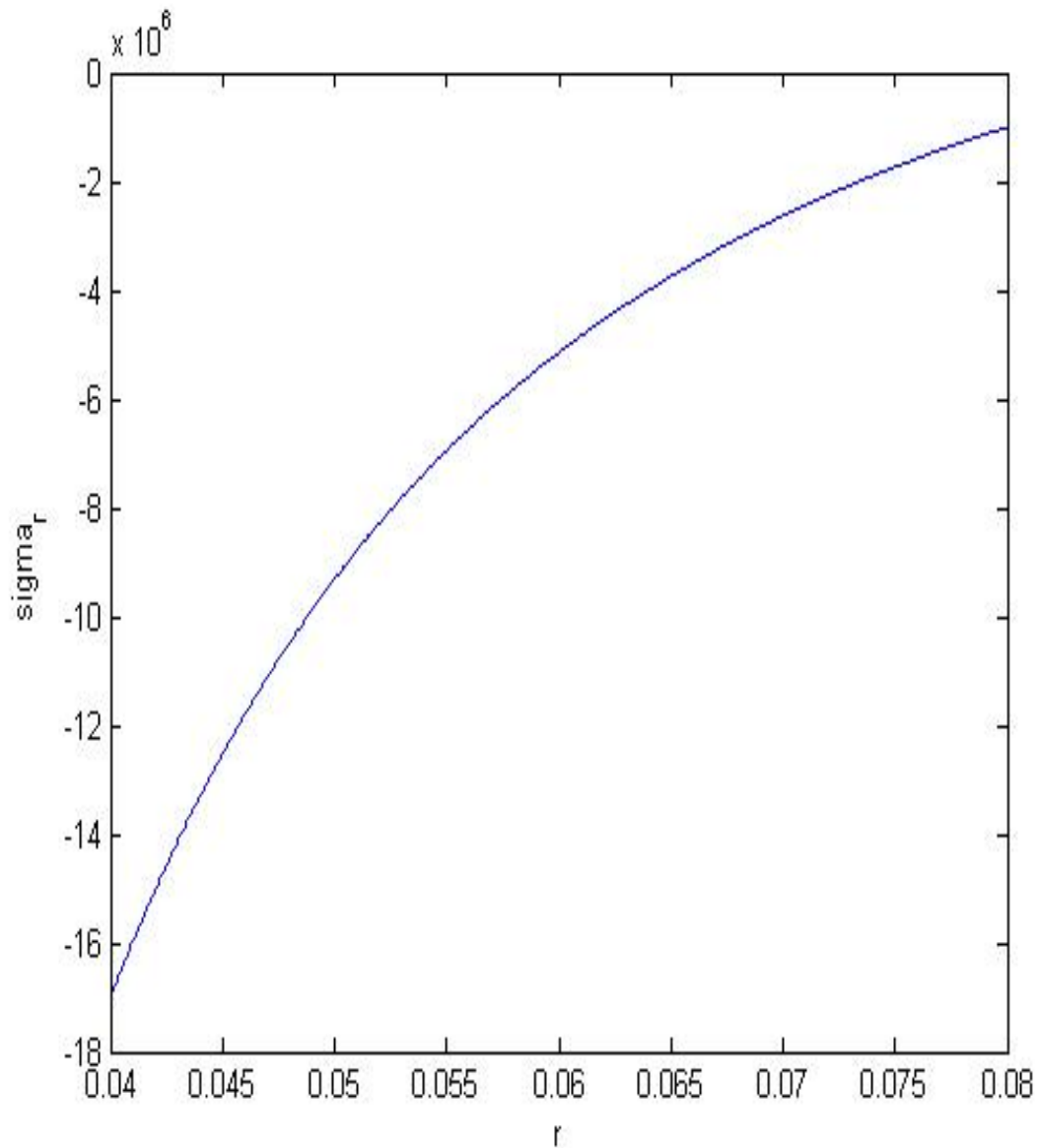


## **RESULTS AND DISCUSSIONS**

**Program for plotting graph between radial stress and radius for thick walled cylinder subjected to internal and external pressure**

```
close all
clear all

pa=17000*10^3;
pb=1000*10^3;
a=0.04;
b=0.08;
r=[a:(b-a)/1000:b];
n=1;
while(n<=1001)
    sigma_r(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) - (((a^2*b^2)/r(n)^2)*((pa-
    pb)/(b^2-a^2)));
    sigma_t(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) + (((a^2*b^2)/r(n)^2)*((pa-
    pb)/(b^2-a^2)));
    n=n+1;
end
plot(r,sigma_t)
xlabel('r')
ylabel('sigma_t')
```



**Fig. 3.1: Variation of radial stress along the radius subjected to internal and external pressure**

The graph above shows the variation of radial stress along the radius of thick walled cylinder subjected to internal and external pressure. The graph shows that radial stress in this case is a compressive stress as its magnitude is negative throughout the graph.

**Program for plotting graph between hoop stress and radius for thick  
walled cylinder subjected to internal and external pressure**

close all

clear all

pa=17000\*10<sup>3</sup>;

pb=1000\*10<sup>3</sup>;

a=0.04;

b=0.08;

r=[a:(b-a)/1000:b];

n=1;

while(n<=1001)

sigma\_r(n)= ((pa\*a<sup>2</sup> - pb\*b<sup>2</sup>)/(b<sup>2</sup> - a<sup>2</sup>)) - (((a<sup>2</sup>\*b<sup>2</sup>)/r(n)<sup>2</sup>)\*((pa-pb)/(b<sup>2</sup>-a<sup>2</sup>)));

sigma\_t(n)= ((pa\*a<sup>2</sup> - pb\*b<sup>2</sup>)/(b<sup>2</sup> - a<sup>2</sup>)) + (((a<sup>2</sup>\*b<sup>2</sup>)/r(n)<sup>2</sup>)\*((pa-pb)/(b<sup>2</sup>-a<sup>2</sup>)));

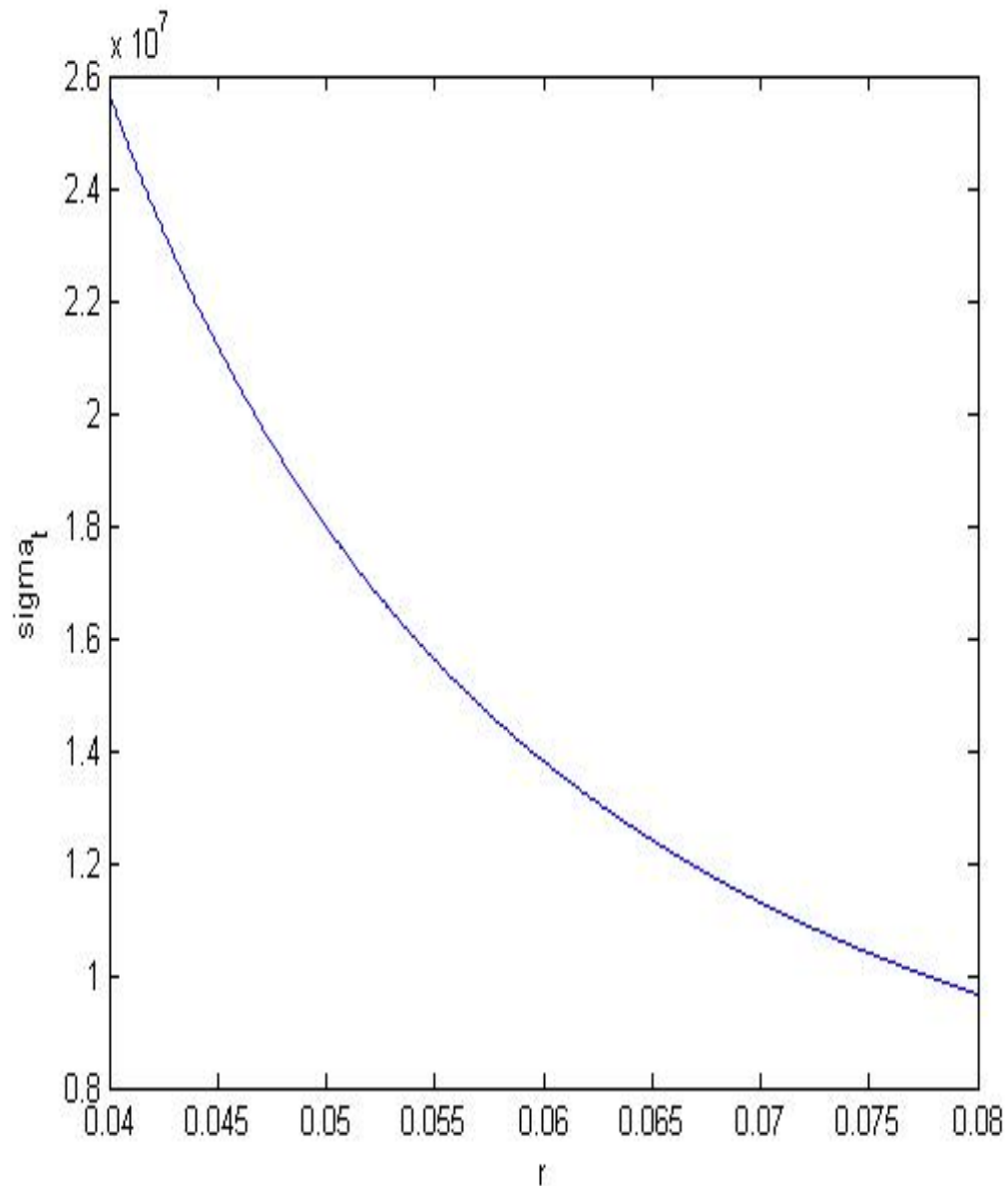
n=n+1;

end

plot(r,sigma\_t)

xlabel('r')

ylabel('sigma\_t')

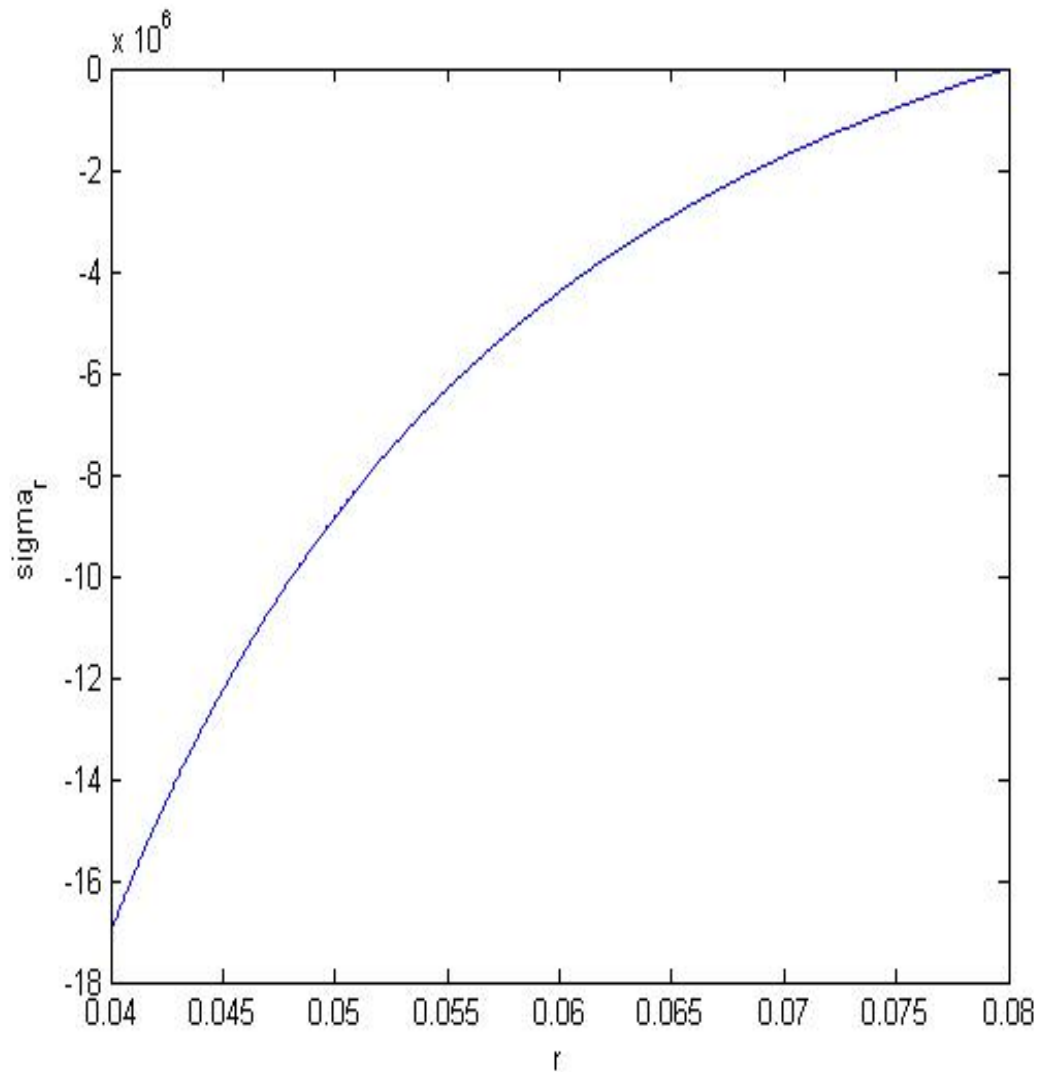


**Fig. 3.2: variation of hoop stress along radius subjected to internal and external pressure**

The graph above shows the variation of hoop stress along the radius of thick walled cylinder subjected to internal and external pressure. The graph shows that hoop stress in this case is a tensile stress as its magnitude is positive throughout the graph.

**Program for plotting graph between radial stress and radius for thick walled cylinder subjected to internal pressure only**

```
close all
clear all
pa=17000*10^3;
pb=0;
a=0.04;
b=0.08;
r=[a:(b-a)/1000:b];
n=1;
while(n<=1001)
sigma_r(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) - (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
sigma_t(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) + (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
n=n+1;
end
plot(r,sigma_r)
xlabel('r')
ylabel('sigma_r')
```



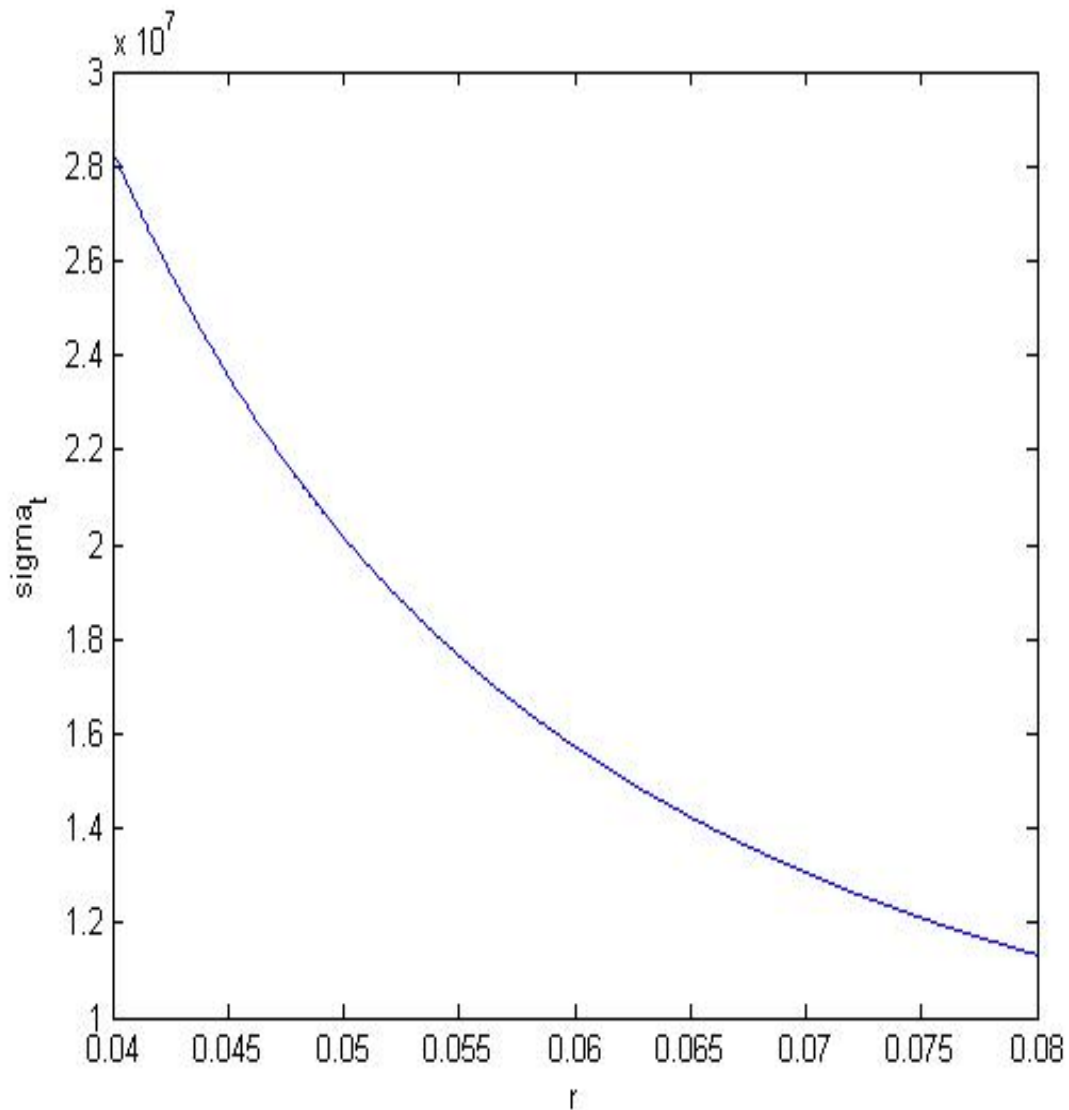
**Fig 3.3: Variation of radial stress along radius subjected to internal pressure only.**

The graph above shows the variation of radial stress along the radius of thick walled cylinder subjected to internal pressure. The graph shows that radial stress in this case is a compressive stress as its magnitude is negative throughout the graph.

**Program for plotting graph between hoop stress and radius for thick  
walled cylinder subjected to internal pressure only**

```
close all
clear all

pa=17000*10^3;
pb=0;
a=0.04;
b=0.08;
r=[a:(b-a)/1000:b];
n=1;
while(n<=1001)
    sigma_r(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) - (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
    sigma_t(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) + (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
    n=n+1;
end
plot(r,sigma_r)
xlabel('r')
ylabel('sigma_r')
```



**Fig 3.4: Variation of hoop stress along radius subjected to internal pressure only.**

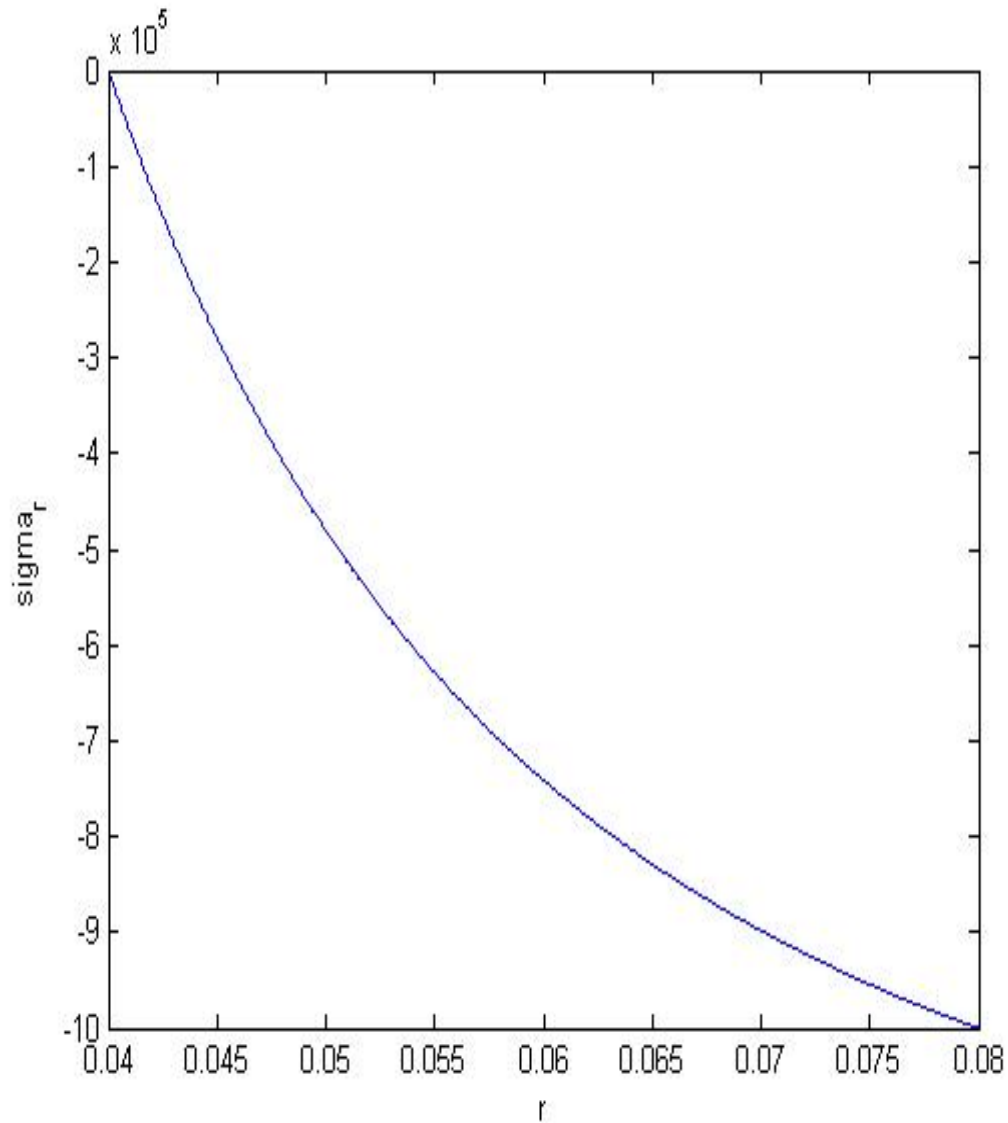
The graph above shows the variation of hoop stress along the radius of thick walled cylinder subjected to internal pressure. The graph shows that hoop stress in this case is a tensile stress as its magnitude is positive throughout the graph.



**Program for plotting graph between radial stress and radius for thick walled cylinder subjected to external pressure only**

```
close all
clear all

pa=0;
pb=1000*10^3;
a=0.04;
b=0.08;
r=[a:(b-a)/1000:b];
n=1;
while(n<=1001)
    sigma_r(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) - (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
    sigma_t(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) + (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
    n=n+1;
end
plot(r,sigma_t)
xlabel('r')
ylabel('sigma_t')
```



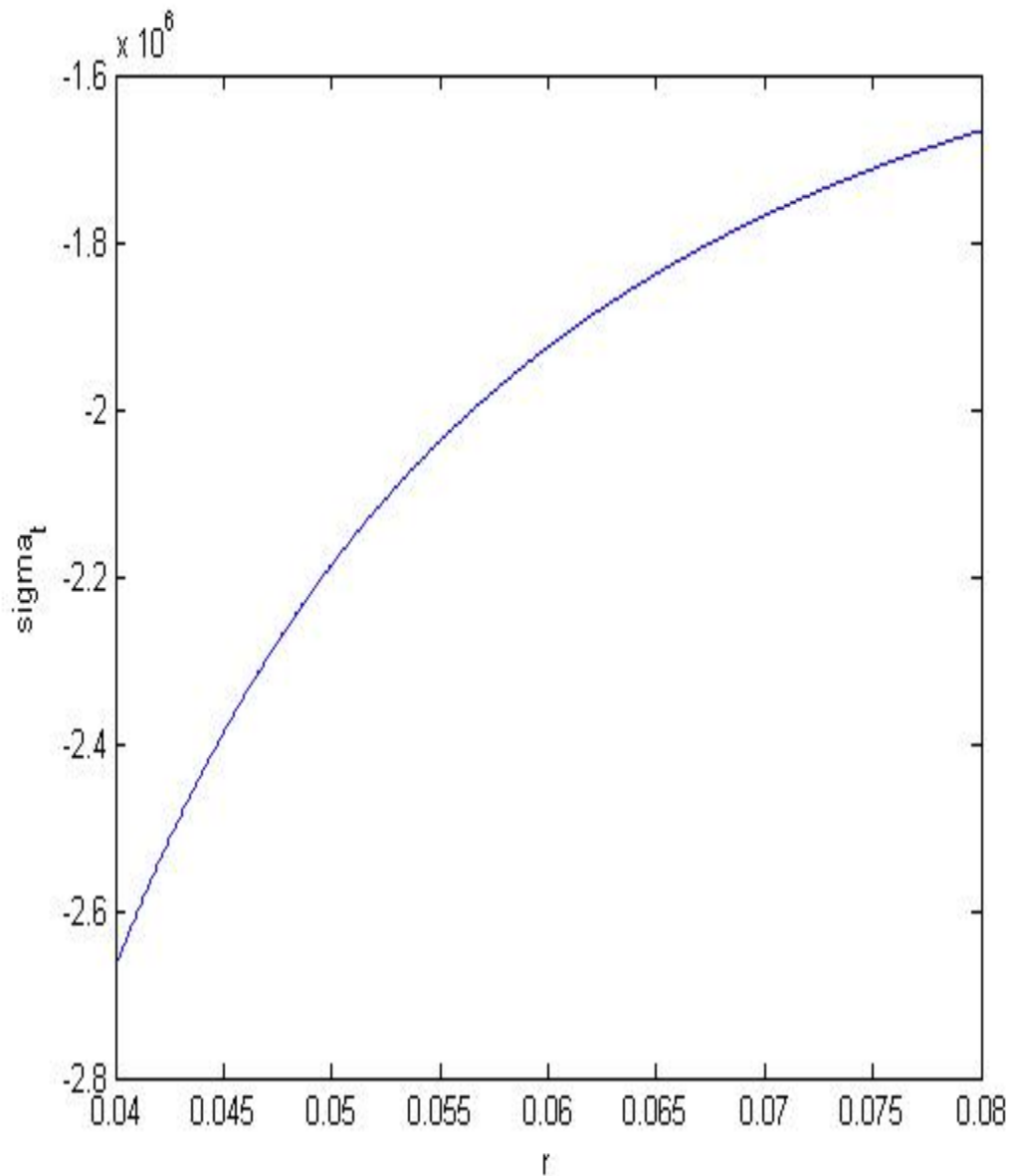
**Fig 3.5: Variation of radial stress along radius subjected to external pressure only.**

The graph above shows the variation of radial stress along the radius of thick walled cylinder subjected to external pressure. The graph shows that radial stress in this case is a compressive stress as its magnitude is negative throughout the graph.

**Program for plotting graph between hoop stress and radius for thick  
walled cylinder subjected to external pressure only**

```
close all
clear all

pa=0;
pb=1000*10^3;
a=0.04;
b=0.08;
r=[a:(b-a)/1000:b];
n=1;
while(n<=1001)
    sigma_r(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) - (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
    sigma_t(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) + (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
    n=n+1;
end
plot(r,sigma_r)
xlabel('r')
ylabel('sigma_r')
```

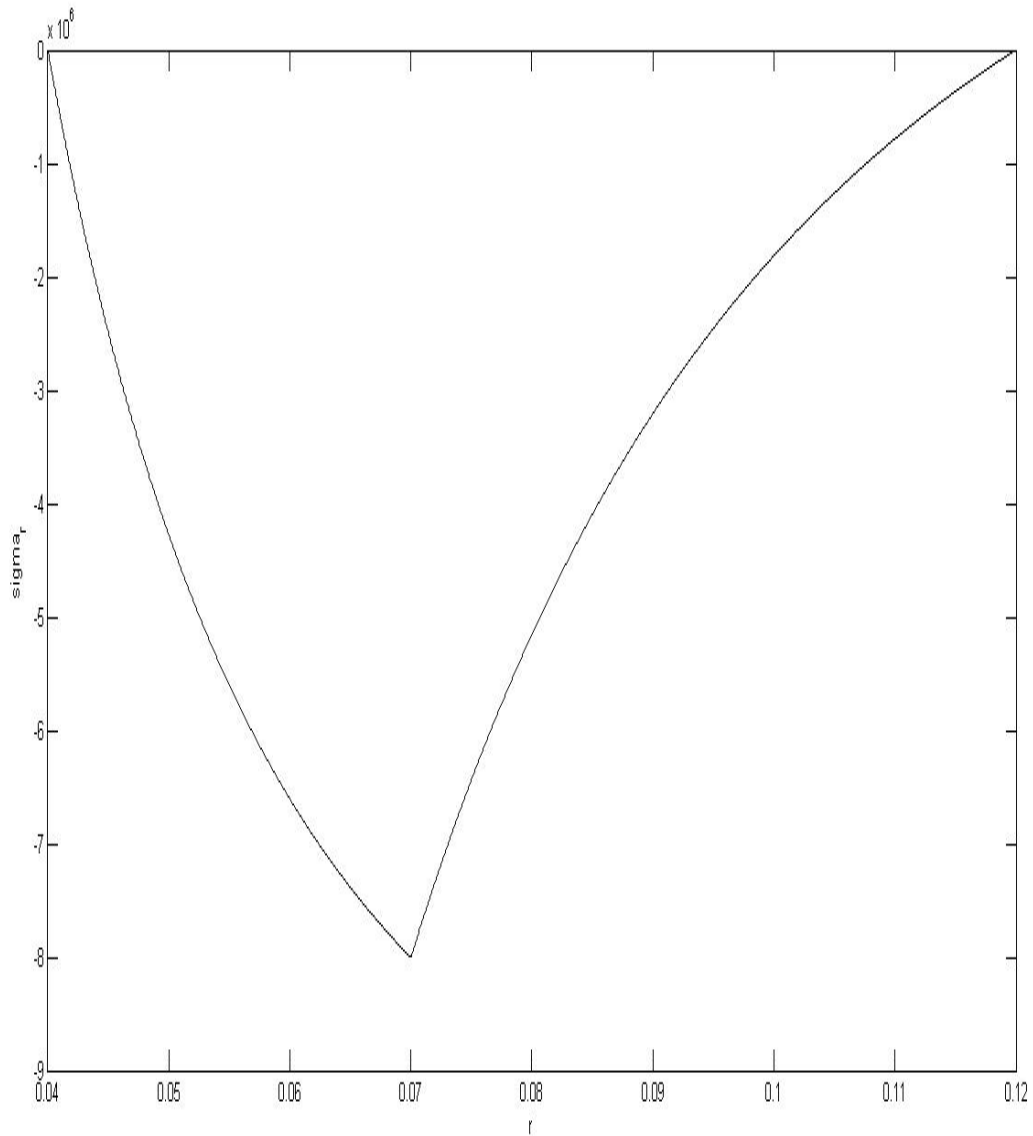


**Fig 3.5: Variation of hoop stress along radius subjected to external pressure only.**

The graph above shows the variation of hoop stress along the radius of thick walled cylinder subjected to external pressure. The graph shows that hoop stress in this case is a compressive stress as its magnitude is negative throughout the graph.

## Program for plotting graph between radial stress and radius for Composite Tubes (Shrink fit)

```
close all
clear all
p=8000*10^3;
a=0.04;
b=0.07;
c=0.12;
r=[a:(c-a)/10000:c];
n=1;
while(r(n)<0.07)
pa=0;
pb=p;
sigma_r(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) - (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
n=n+1;
end
n=n-1;
while(n<=10001)
pa=p;
pb=0;
sigma_r(n)= ((pa*b^2 - pb*c^2)/(c^2 - b^2)) - (((b^2*c^2)/r(n)^2)*((pa-
pb)/(c^2-b^2)));
n=n+1;
end
plot(r,sigma_r,'k')
xlabel('r')
ylabel('sigma_r')
```

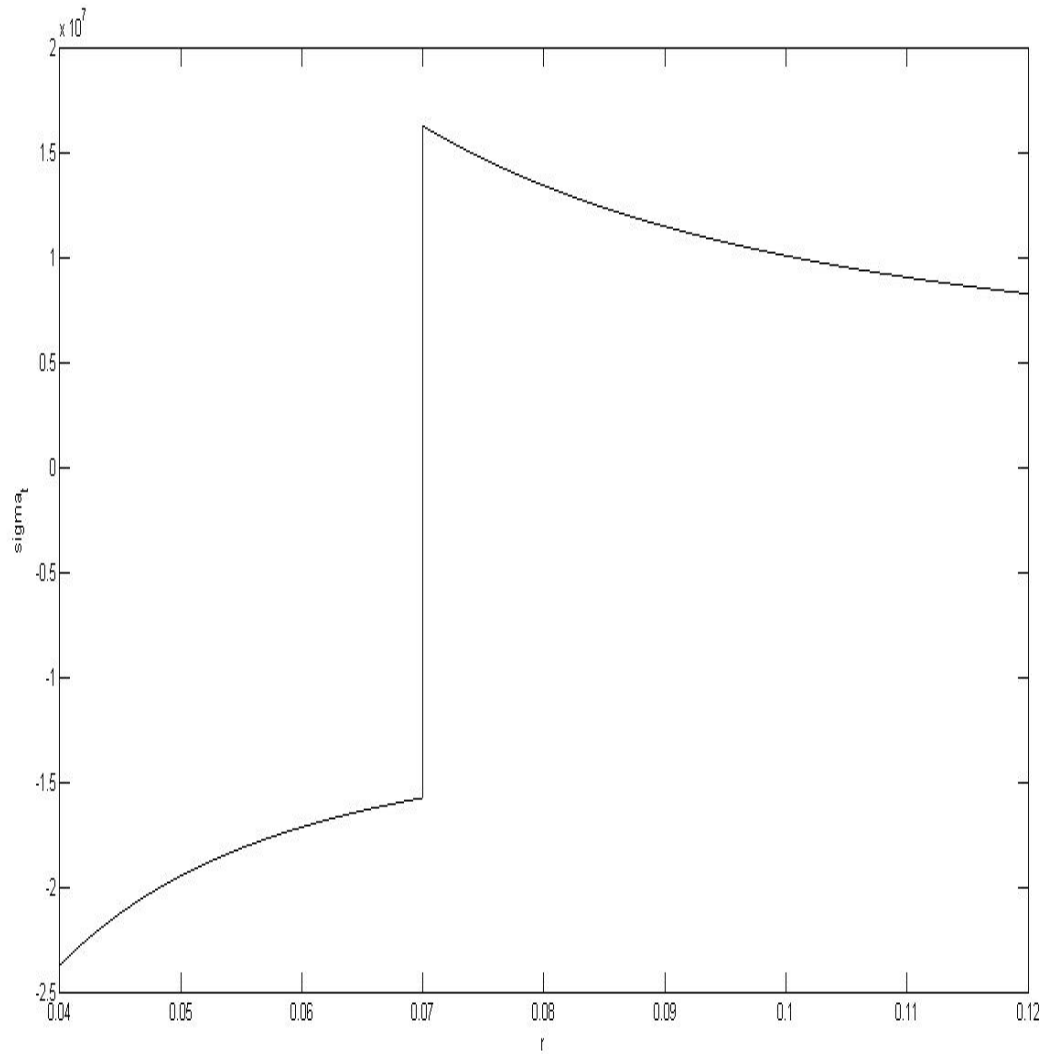


**Fig. 3.7: Variation of radial stress along radius in case of shrink fit**

The graph above shows the variation of radial stress along the radius in case of shrink fit. The graph shows that radial stress in this case is a compressive stress as its magnitude is negative throughout the graph. It can also be seen that the magnitude of radial stress increases upto the external radius of the inner radius and then decreases till it reaches zero at the external radius of the outer cylinder.

**Program for plotting graph between stress and radius for Composite  
Tubes (Shrink fit)**

```
close all
clear all
p=8000*10^3;
a=0.04;
b=0.07;
c=0.12;
r=[a:(c-a)/10000:c];
n=1;
while(r(n)<0.07)
pa=0;
pb=p;
sigma_t(n)= ((pa*a^2 - pb*b^2)/(b^2 - a^2)) + (((a^2*b^2)/r(n)^2)*((pa-
pb)/(b^2-a^2)));
n=n+1;
end
n=n-1;
while(n<=10001)
pa=p;
pb=0;
sigma_t(n)= ((pa*b^2 - pb*c^2)/(c^2 - b^2)) + (((b^2*c^2)/r(n)^2)*((pa-
pb)/(c^2-b^2)));
n=n+1;
end
plot(r,sigma_t,'k')
xlabel('r')
ylabel('sigma_t')
```



**Fig. 3.7: Variation of hoop stress along radius in case of shrink fit**

The graph above shows the variation of hoop stress along the radius in case of shrink fit. The graph shows that hoop stress in this case is a compressive stress as its magnitude is negative upto the external radius of the inner cylinder and is a tensile stress from thereon till the external radius of the outer cylinder as its magnitude is positive.



## CHAPTER 4

### **4.1 Summary**

An attempt has been made to know the load capacity of a thick walled cylinder . Classical bookwork formulas have been employed to obtain the stress distribution in cylinder subjected to internal and external pressure. Variation of stress across the thickness are shown using Matlab.

### **4.2 Future Scope of Work**

In this Project, Stress analysis is done on simple thick walled cylinder. But in industries there is wide application of thick walled cylinder having holes. Stress analysis of that should be done in future. Plasticity and yield strength of material should be analysed so that we can have better understanding while using. In case of shrink fit there was no external forces were applied in analysis. In future it should be analysed with external forces

## **REFERENCES**

- [1] S. Xu & M. Yu, “Shakedown Analysis of thick walled cylinders subjected to internal pressure with unified strength criterion”, International Journal of Pressure vessels & piping, Vol. 82, pp706-712, 2005
- [2] M. H. Hojjati & A. Hossaini, “Theoretical & Finite Element Modelling of autofrettage process in strain hardening thick walled cylinders”, International Journal of Pressure vessels & piping, Vol. 84, pp 310-319, 2007
- [3] M. Imaninejad & G. Subhash, “proportional loading of thick walled cylinders”, International Journal of Pressure vessels & piping, vol. 82, pp129-135, 2005
- [4] Y. Z. Chen & X. Y. Lin, “An alternative numerical solution of thick walled cylinders and spheres made of functionally graded materials”, Computational Materials Science, vol. 48, pp640-647, 2010
- [5] L. Li & M. Anburton, “Anelasto-plastic evaluation of the stress state around cylindrical openings based on a closed multiaxial yield surface”, International Journal of Pressure vessels & piping, vol. 33, pp193-213, 2009.